Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)
ISSN 1673-565X (Print); ISSN 1862-1775 (Online)
www.zju.edu.cn/jzus; www.springerlink.com
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Experimental study on a −60 °C cascade refrigerator with dual running mode*

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Received Apr. 18, 2011; Revision accepted Dec. 31, 2011; Crosschecked Apr. 19, 2012

Abstract: In this study, an experimental setup is designed and built to investigate the feasibility and performance of the proposed dual-mode cascade refrigeration cycle. The apparatus can be operated in two modes: dual-stage mode and single-stage mode such that the low temperature cycle (LTC) can be operated together with the high temperature cycle (HTC) or can run independently. Experimental results validate the feasibility of independent operation of LTC. The performance of the independent operation of LTC mode is theoretically investigated using experimental data as bases. Detailed suggestions are made for the improvement of the coefficient of performance (COP) of the experimental system. Theoretically, high COP of the cycle provides excellent application for the presented refrigeration cycle.

Key words: Cascade refrigeration, Dual-mode, Regenerator, Optimization, Effectiveness
doi:10.1631/jzus.A1100107 **Document code:** A **CLC number:** TB657.3

1 Introduction

The cascade refrigeration cycle, which can yield a large temperature difference between the heat source and heat sink, is widely employed in industrial applications to achieve temperatures that are considerably higher or lower than ambient temperature. It is also used to satisfy simultaneous heating and cooling requirements. Cascade systems are typically applied in temperature ranges of −70 to 100 °C (Dincer, 2003; Wang *et al*., 2009; 2011). Recent studies on cascade refrigeration cycles have focused mainly on the use of natural refrigerants and optimization of the cascade refrigeration-heat pump system. For example, some studies have been carried out on the performance and application of the NH_3-CO_2 cascade refrigeration system (Belozerov *et al*., 2007; Alberto Dopazo *et al*., 2009; Wang *et al*., 2009). Sarkar *et al*. (2004) and Bhattacharyya *et al*. (2005; 2007; 2008) studied the CO_2-C_3H_8 cascade refrigeration-heat pump system. Generally, the focuses of the aforementioned studies are the cycle itself and alternative refrigerant properties. New designs, however, may promote the use of the cascade refrigeration cycle and improve system reliability and stability.

Taking advantage of the characteristics of an internal heat regenerator, we propose an improved cascade refrigeration cycle to investigate the independent operation of the low temperature cycle (LTC). A fully-instrumented test rig is built to evaluate performance. The main focus of this study is to investigate the feasibility of the proposed system and the effect of operating conditions on system performance when LTC operates independently.

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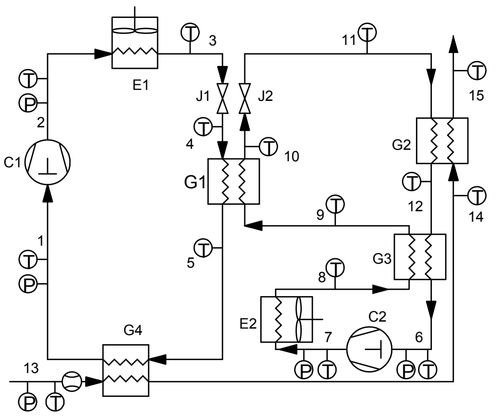
* Project supported by the National Natural Science Foundation of China (No. 50976082), the Specialized Research Fund for the Doctoral Program of Higher Education (No. 20090201110006), and the State Key Laboratory of Multiphase Flow of Xi'an Jiaotong University, China

2 Experimental

2.1 Flow loop

Three main parts make up the proposed cascade refrigeration system with dual running mode (Fig. 1): the high temperature cycle (HTC) (1-2-3-4-5-1) with CHF_2Cl as the working fluid, the LTC (6-7-8-9-10-11-12-6) with CHF_3 as the working fluid, and the load pipe (13-14-15). HTC consists of compressor C1, condenser E1, throttling element J1, condenser-evaporator G1, and gas cooler G4. LTC consists of compressor C2, condenser E2, internal regenerator G3, condenser-evaporator G1, throttling element J2, and gas cooler G2. HTC and LTC are connected by condenser-evaporator G1. A load of dry air is cooled down when it flows through gas coolers G4 and G2 successively (13 to 15 in Fig. 1).



Flow meter Temperature sensor Pressure transmitter
C1: compressor of HTC; C2: compressor of LTC;
E1: condenser of HTC; E2: condenser of LTC;
J1: throttle of HTC; J2: throttle of LTC;
G1: condenser-evaporator; G2: evaporator of LTC;
G3: regenerator of LTC; G4: gas cooler of HTC

Fig. 1 Flowchart of the improved two-stage cascade refrigeration cycle

This test apparatus can be operated in two modes with an internal heat regenerator and a condenser employed in LTC, i.e., LTC can be operated alone or with HTC. This makes the system unique when compared with the conventional dual-stage cascade cycle. The refrigerant from compressor C2 is cooled by condenser E2, and then further cooled by internal heat exchanger G3 through the cold energy recycled from the evaporator outlet. When inlet temperature 9 approaches outlet temperature 10 of the condenser-evaporator, the cooling effect of the condenser-evaporator becomes negligible, and LTC can be operated independent of HTC to achieve the same evaporation temperature value.

The improved system can be employed in small load conditions, such as cryostat. It possesses the following beneficial advantages such that the refrigerant is heated by internal heat exchanger G3 before it enters compressor C2, yielding a better oil return for the compressor and ensuring the stability of LTC. For minimal demands on cold energy, LTC can satisfy requirements alone having the aid of HTC at start-up. Thus, the system does not require frequent start-ups, and only one compressor needs to work after operation has been initiated. Such attributes present energy-savings.

2.2 Determination of state points

Design parameters such as the isentropic efficiencies of the compressors, overlap temperature of the system, temperature difference, etc., significantly affect the coefficient of performance (COP) of a system. Thus, a parametric study is indispensable for optimizing the design parameters and determining all state points. The following assumptions are made to simplify the thermodynamic analysis:

1. The compression processes are adiabatic but not isentropic. An isentropic efficiency of 0.8 is assumed for both the HTC and LTC compressors.

2. Pressure drops in connecting pipes, compressor valves, and heat exchangers are negligible.

3. The ambient temperature is 25 °C. Refrigerants at the condenser outlet of HTC are saturated; a temperature difference of 5 °C exists between the HTC condensing temperature and the ambient temperature.

4. Regenerator effectiveness is 0.8.

5. The overlap temperature in the condenser-evaporator is 5 °C.

6. The subcooling degree before the throttling element is 5 °C.

7. The evaporating temperature of LTC is −65 °C.

To facilitate the analysis, we have developed an exclusive computer code for the thermodynamic properties of the modified cascade cycle. The state parameters of the cycle are determined according to the design requirements and principles of system performance optimization (Table 1).

Table 1 Design parameters for all state points of the cooling cycle

State point	Temperature (°C)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/(kg·K))
1	−33	0.115	392.06	1.83
2	95	1.192	468.94	1.87
3	30	1.192	236.62	1.13
4	−38	0.115	236.62	1.17
5	−38	0.115	290.20	1.40
6	15	0.247	388.25	1.93
7	95	0.904	448.22	1.96
8	30	0.904	393.16	1.80
9	−33	0.904	268.16	1.30
10	−38	0.904	144.35	0.79
11	−65	0.247	144.35	0.79
12	−65	0.247	263.25	1.37

2.3 Experimental setup

Fig. 2 shows the experimental setup of the dual-mode cascade refrigeration system. Two commercial hermetic rolling piston air-conditioning compressors C1, C2 of model YZG-53RB (Qing'an Refrigeration Equipment Co., Ltd., Xi'an, China), two air-cooled finned tube condensers E1, E2, four customized SCE (Aotaihua Refrigeration Equipment Co., Ltd., Beijing, China) model coaxial-pipe heat exchangers G1, G2, G3, and G4, and two capillaries J1, J2 are employed in the experimental equipment. Effective insulation measures are taken to reduce cold loss. The components and pipelines were wrapped in polyethylene foam and radiation-shield aluminum foil to reduce cold loss through conduction and radiation, respectively.

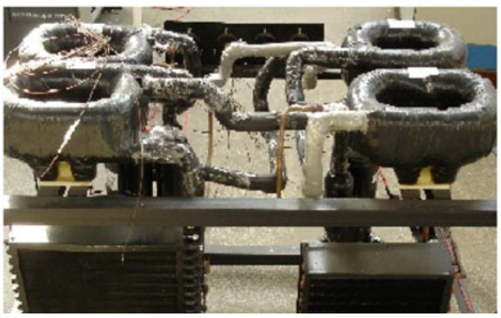


Fig. 2 Experimental apparatus of the dual-mode cascade refrigeration cycle

2.4 Data acquisition system

The temperatures and pressures of the refrigerant are measured at various locations in the experimental equipment (Fig. 1). Copper-constantan thermocouples and pressure/vacuum gauges are used to measure the temperatures and the pressures of each state point, respectively. A glass rotameter is used to measure the flow rate of the cooled air. Cooling capacity is estimated by measuring the flow rate, as well as the inlet and outlet temperatures of the cooled air.

Data acquisition system of the experimental apparatus is composed of hardware and software. The hardware part includes an industrial personal computer, PCI CEC488GPIB (Keithley Instruments, Inc., Cleveland, USA) general interface card, thermocouple temperature sensor, Keithley model 2000 digital multimeter, Keithley 7001 switching system, and 7014 collecting card. A set of experimental data acquisition system based on Labview platform is developed for the real time data collection and observation, by which the programmable logic controller (Advantech Co., Ltd., Taiwan, China) initialization, channel configuration and collecting command dispatch for real time data displaying and storage being realized.

3 Results and discussion

The experimental process can be divided into two stages. The first is the dual-stage mode, in which HTC begins to operate first, followed by LTC after HTC has achieved a steady run. The second is the single LTC stage mode, in which LTC is switched on after HTC has been running steadily. HTC is then switched off after LTC has achieved a steady run. This enables LTC to run independently.

Fig. 3 shows the operating condition of the dual-stage mode. Before initialization of LTC, HTC is allowed to run until it has reached a stable operating condition. LTC is initiated about 85 min later. As the experimental apparatus is equipped with G1, the throttle valve outlet temperature of LTC is cooled down. Because of the heat load of LTC, the loaded air outlet temperature of LTC is higher than that of HTC. After LTC is initiated, the throttle valve outlet temperature and loaded air outlet temperature of LTC decrease dramatically. LTC tends to stabilize after a small increase in the throttle valve outlet temperature and air outlet temperature of HTC. With airflow rate of 5.35 g/s, the loaded air is cooled to −61 °C and the cooling capacity of the system is about 401 W.

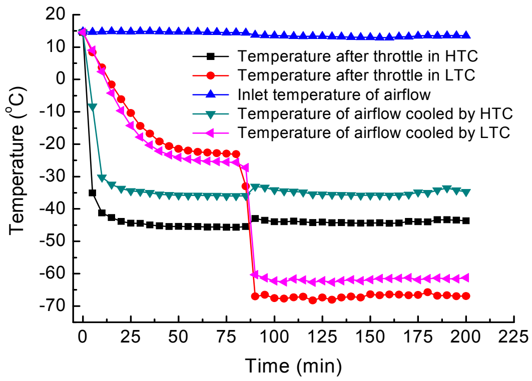


Fig. 3 Variations in temperature of the two-stage cascade cycle

Fig. 4 shows the operating condition of the single LTC stage mode. LTC begins about 85 min after the initiation of HTC, while HTC is switched off after 100 min, and LTC begins to run independently. LTC works independently for about 100 min and the throttle valve outlet temperature of LTC is maintained at about −60 °C.

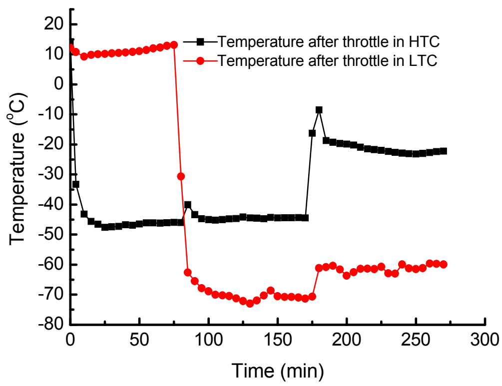


Fig. 4 Variations in temperature during independent operation of LTC

In the tests, the thermocouple temperature sensor has an accuracy of ±0.2 °C in the temperature range from −70 to 100 °C. The accuracy of pressure sensor used in the experiment is ±1.6% in the pressure range from −100 to 3500 kPa. The accuracy of glass rotameter is ±2.5% in the range from 0.6 to 6 m^3/h. The uncertainty analysis on the cooling capacity was performed according to the error propagation equation (Kline and Mcclintock, 1953). The uncertainty of the cooling capacity is about ±4%.

4 System optimization

4.1 Effects of regenerator effectiveness and condensing temperature on LTC performance

The internal heat regenerator and condenser employed in LTC are the major distinctions between this cycle and the conventional dual-stage cascade cycle. The performance of each component would have a decisive effect on the independent operation of LTC. At the same time, the specific refrigeration capacity of the LTC evaporator has a direct influence on system performance. The effectiveness of the regenerator (ε) is defined as the ratio of the actual rate of heat transfer (φ) to the maximum possible rate of heat transfer ($\varphi_{\max}$):

$$\varepsilon = \frac{\varphi}{\varphi_{\max}}, \tag{1}$$

$$\varphi_{\max} = \min\{\varphi_1, \varphi_2\}, \tag{2}$$

where

$$\begin{aligned}\varphi_1 &= \dot{m}_{\mathrm{H}}(h_{t'_{\mathrm{H}},p'_{\mathrm{H}}} - h_{t'_{\mathrm{C}},p'_{\mathrm{H}}}) \\ &= \dot{m}_{\mathrm{H}}(355.19 - 106.53) = 248.66\dot{m}_{\mathrm{H}},\end{aligned} \tag{3}$$

$$\begin{aligned}\varphi_2 &= \dot{m}_{\mathrm{C}}(h_{t'_{\mathrm{H}},p'_{\mathrm{C}}} - h_{t'_{\mathrm{C}},p'_{\mathrm{C}}}) \\ &= \dot{m}_{\mathrm{C}}(389.28 - 159.86) = 229.42\dot{m}_{\mathrm{C}},\end{aligned} \tag{4}$$

where ε is the effectiveness, φ is the heat transfer rate, $\dot{m}$ is the mass flow rate, t is the temperature, p is the pressure, and h is the specific enthalpy. Subscripts H and C denote hot fluid and cold fluid, respectively, and superscripts ′ and ″ refer to the inlet and outlet, respectively. According to the experimental data, $t'_{\mathrm{H}} = 17$ °C, $t'_{\mathrm{C}} = -65$ °C, $p'_{\mathrm{H}} = 2800$ kPa, and $p'_{\mathrm{C}} =$ 280 kPa.

The values of $h_{t'_{\mathrm{H}},p'_{\mathrm{H}}}$, $h_{t'_{\mathrm{C}},p'_{\mathrm{H}}}$, $h_{t'_{\mathrm{H}},p'_{\mathrm{C}}}$, and $h_{t'_{\mathrm{C}},p'_{\mathrm{C}}}$ are calculated using Eqs. (3) and (4).

As $\dot{m}_{\mathrm{H}} = \dot{m}_{\mathrm{C}}$:

$$\varphi_{\max} = \min\{\varphi_1, \varphi_2\} = \varphi_2, \tag{5}$$

and

$$\varphi = \dot{m}_{\mathrm{H}}(h_{t'_{\mathrm{H}},p'_{\mathrm{H}}} - h_{t''_{\mathrm{H}},p''_{\mathrm{H}}}). \tag{6}$$

Substituting Eqs. (5) and (6) into Eq. (1), ε can be calculated as

$$\varepsilon = \frac{\varphi}{\varphi_{\max}} = \frac{\dot{m}_{\mathrm{H}}(355.19-158.43)}{229.42\dot{m}_{\mathrm{H}}} = 85.75\%. \tag{7}$$

When the cooling temperature is −60 °C, the compressor has an isentropic efficiency of 0.8 and the pressures maintain the experimental values. Pressure loss is negligible as assumed, and the amount of cold lost of the condenser-evaporator is maintained at the same value. Fig. 5 depicts the variations in system COP with regenerator effectiveness and specific refrigeration capacity of the LTC evaporator when the temperature of the condenser outlet is kept at 25 °C. Maintaining regenerator effectiveness at 85.75%, the variations in system COP affected by variations in condensing temperature and specific refrigeration capacity are presented in Fig. 6.

As shown in Fig. 5, the COP increases with an increase in specific refrigeration capacity, which consequently demands high effectiveness for the regenerator. For instance, when the specific refrigeration capacity is 30 kJ/kg, LTC can run independently only when the effectiveness of the regenerator is higher than 96%. This is because more substantial refrigeration capacity release entails less refrigeration capacity that may be used for regeneration. The regenerator should achieve higher effectiveness to obtain the required temperature before throttling. Otherwise, the temperature after throttling fails to reach the required temperature because of obtaining the high temperature before throttling. Thus, to improve the stability and COP of the LTC, effective heat exchanger such as plate-fin heat exchanger should be used as a regenerator.

Fig. 6 illustrates that the lower the condensing temperature is, the more substantial the specific refrigeration capacity released, and the higher COP is attained. This experiment is carried out in winter, so that the condensing temperature can be lowered to 25 °C through air-cooling. In many conditions, however, air-cooled condensers cannot reach such a low temperature; hence, a water-cooled condenser should be employed.

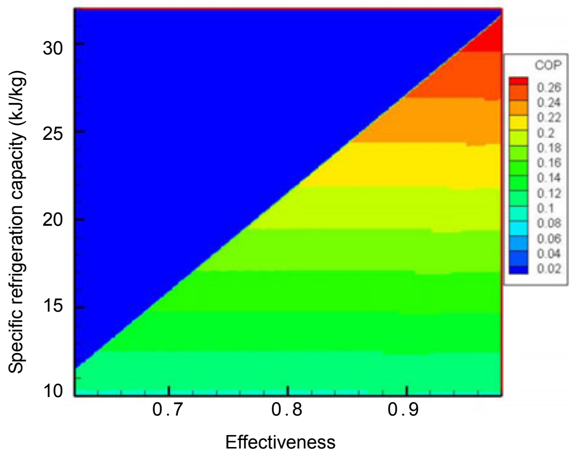


Fig. 5 Variations in system COP with efficiency of regenerator and specific refrigeration capacity

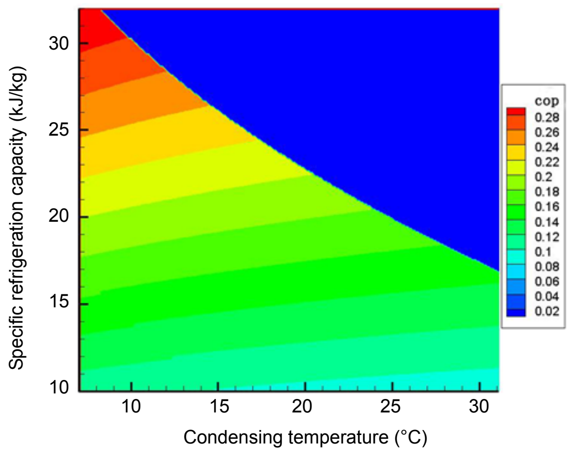


Fig. 6 Variations in system COP with condensing temperature and specific refrigeration capacity

4.2 Effects of pressure ratio and condenser-evaporator temperature difference on LTC performance

Theoretically, system COP can attain its maximum for a particular value of pressure ratio. In search of an optimum pressure ratio, pressure loss is assumed to be negligible and low pressure is maintained at 280 kPa. The compressor has an isentropic efficiency of 0.8, and the condensing temperature is maintained at 25 °C. A corresponding thermodynamics calculation model is established, and an exclusive computer code is developed for the thermodynamic properties of LTC. The variation in system COP with pressure ratio and temperature difference in the condenser-evaporator is presented in Fig. 7.

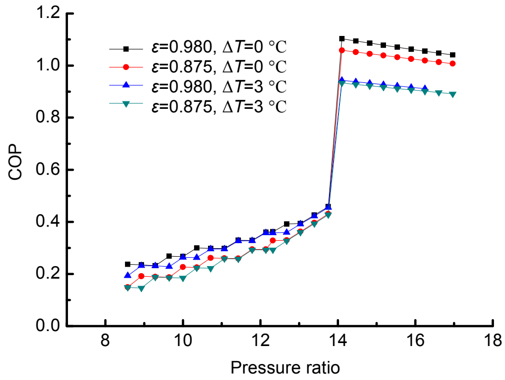


Fig. 7 Variations in system COP with pressure ratio and temperature difference in the condenser-evaporator

As shown in Fig. 7, system COP first increases slowly, while pressure ratio is at around 14.0, COP sharply increases to a maximum, and then gradually decreases. This result can be attributed to the assumption that the condensing temperature remains at 25 °C, and the condenser releases more heat energy with the increase in pressure ratio. When the pressure ratio reaches 14.0, the refrigerant is cooled from gaseous to liquid in the condenser and consequently a mass of latent heat of liquefaction is discharged causing a sudden increase in system COP. When the pressure ratio increases further, the compressor power consumption increases. The heat discharged by the condenser grows slightly, thereby diminishing system performance. A commercial air-conditioning compressor was used in this system as a high-pressure ratio compressor is needed to improve system COP. Because of the substantial increase in heat energy discharge, the common air-cooled condenser is not expected to satisfy the requirements. A plate-fin condenser with high heat transfer effectiveness should therefore be adopted (Chen *et al.*, 2008). We also find that system COP improves by about 20% when there is no temperature difference in the condenser-evaporator. Adding a bypass between the inlet and outlet of the condenser-evaporator is effective. The bypass is connected when the low-temperature cycle runs alone; otherwise, it is removed.

The analysis of the results confirms that the heat regenerator, condenser, and compressor are the key components for the independent operation of LTC. When the appropriate components are employed, system COP can generate a value of about 1.1 while LTC is running independently. This is efficient for the low refrigeration temperature of −60 °C.

5 Conclusions

A cascade refrigeration cycle with a dual running model is presented. The experimental study is carried out, and a theoretical model is established for the performance evaluation of the cycle. The following conclusions can be drawn:

1. Preliminary experimental investigation indicates that this dual-mode cascade refrigeration cycle system can be operated in two modes. LTC can realize independent operation only without load. Though system performance is a bit poor, system feasibility and stability are improved because only one compressor is needed when LTC runs independently.

2. The heat regenerator, condenser, and compressor are essential to the performance of LTC. Methods for improving COP include: increasing the effectiveness of the regenerator, optimizing compression ratio, enhancing the heat exchange in the condenser, and adding a bypass between the inlet and outlet of the condenser-evaporator.

3. The theoretical calculation of the COP of the improved cycle is as high as 1.1 at the refrigeration temperature of −60 °C, indicating that the dual-mode cycle for a low temperature refrigerator is feasible and holds potential.

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